COMPUTER AIDED INTEGRATED DESIGN OF GEAR SPEED REDUCERS

by

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INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

APRIL, 1989

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OF
GEAR SPEED REDUCERS

A Thesis Submitted
in Partial Fulfilment of the Requirements
for the Degree of

MASTER OF TECHNOLOGY

by

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to the

DEPARTMENT OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

APRIL, 1989

C E R T I F I C A T E

This is to certify that the thesis entitled "COMPUTER AIDED INTEGRATED DESIGN OF GEAR SPEED REDUCERS", by Abhay M. Tarnekar, has been completed under my supervision and that this work has not been submitted elsewhere for a degree.

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March 1989

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ACKNOWLEDGEMENTS.

I sincerely wish to thank Professor B.Sahay who helped me through all stages of the work. This work would not have been realised without his able guidance.

I also thank all others who have helped me towards the completion of the work in a direct or indirect way.

Abhay.

ABSTRACT

Gear speed reducers are essential components of any mechanical system in which angular speed reduction takes place by means of gears. Designing such a system, which includes numerous machine elements, requires substantial amount of time and efforts. It becomes even more time consuming if the design is to be reprocessed again with change in the value of some parameters.

In the present work, computer aided integrated design of gear speed reducer with individual components design has been presented. Spur, Helical (Herringbone) and Bevel gears are included as the basic gear elements. The implemented system is interactive and user-friendly in nature. It can serve as a design tool for machine designers. Starting from the design of gears, shafts, etc., the layout of these elements with their dimensions can be obtained.

C O N T E N T S

LIST OF SYMBOLS

Design of Gears

$[M_t]$	Design twisting moment kg-cm
$[\sigma_b]$	Design bending stress kg/sq.cm
$[\sigma_c]$	Design surface compressive stress kg/sq.cm
$[\sigma_{-1}]$	Design endurance stress in bending kg/sq.cm
E	Equivalent Young's Modulus kg/sq.cm
BHN	Brinell's Hardness Number of the material
σ_u	Ultimate strength kg/sq.cm
σ_y	Yield strength kg/sq.cm
n	Factor of safety in bending
σ_c	Induced surface compressive stress kg/sq.cm
σ_b	Induced bending stress kg/sq.cm
q	Unit load kg/cm
m	Standard module cm
m_n	Standard normal module cm
m_t	Transverse module cm
m_{av}	Average module cm
Z	Number of teeth
Z_v	Virtual number of teeth
a	Center distance cm
b	Face width cm
R	Cone distance cm
ψ_y	R / b
ψ	b / a
ψ_m	b / m
δ	Reference angle for bevel gears
d	Pitch circle diameter cm
$\emptyset$	Pressure angle
β	Helix angle
k_{bl}	Life factor for bending
k_{cl}	Life factor for surface strength

k_σ	Stress concentration factor
k_d	Dynamic load factor
k	Load concentration factor

Bearing Selection

P_a	Axial load kg
P_r	Radial load kg
P_{eq}	Equivalent load kg
X	Radial factor
Y	Axial factor
S	Service factor
C	Dynamic capacity kg
C_0	Static capacity kg
L	Required life of bearing in million revolution (mr)
L'_{10}	Calculated life of bearing for p_{10} in mr, rating life
e	P_a / P_r
p	Probability of survival of the given bearing
p_{10}	Probability of survival = 0.90 (90 %)

Design of Keys

w	Width of key cm
t	Thickness of key cm
l	Length of key cm
d	Reference diameter of shaft
$[\tau]$	Design shear stress kg/sq.cm
$[\sigma_c]$	Compressive stress of the material kg/sq.cm

Design of Shafts

P_a	Axial load on shaft kg
α	Column action factor

d	Diameter of shaft cm
K_b	Combined shock and fatigue factor applied to M_b
K_t	Combined shock and fatigue factor applied to M_t
l	Length of the shaft cm
M_t	Twisting moment kg-cm
M_b	Bending moment kg-cm
$[\tau]$	Design shear stress kg/sq.cm
τ	Shear stress kg/sq.cm
σ_{-1}	Endurance limit stress of the shaft material kg/sq.cm

Design of Sleeves

t	Thickness of sleeve cm
P_a	Axial load acting kg
D_{inner}	Inner diameter of sleeve cm
$[\sigma_c]$	Design compressive stress kg/sq.cm
σ_c	Induced compressive stress kg/sq.cm

C H A P T E R   1

INTRODUCTION

1.1 INTRODUCTION

Transmission of torque between two bodies in mutual contact with uniform angular velocity can be achieved in two basic ways. One consists of allowing these to touch each other with some contact force. The torque is transmitted because of the frictional force existing between them. This type of drive is known as friction drive. The basic drawback with these drives is that the possibility of slip cannot be avoided also, this slip can vary with the change in load; and the angular velocity ratio cannot remain constant if torque is varying. The other way is to use positive drives. These drives ensure a positive contact. On account of this fact, slip is absent in these drives, irrespective of torque magnitude. Geared drives come under this category. The power is transmitted because of mutual engagement of gear teeth. The geometry of tooth decides the smoothness of operation. This drive is more complex to design but it is indispensable when a constant angular velocity ratio is to be maintained. The present work is on angular speed reduction by means of geared drives.

The term "Gear speed reducer" doesn't require much clarification. It is a mechanical system, capable of reducing the angular speed by means of a gear train. It can contain any type

of gear for the speed reduction, depending upon the working conditions and the need of problem. These gears can be mounted either on parallel or non parallel shafts. The total speed reduction can be achieved in several stages, depending on requirement. However, the speed ratio of each stage is fixed and cannot be changed during the operation.

The speed reducer is necessary when heavy torque is required at a lower speed. A typical example is a "ball mill", used primarily in cement industries, which works at a much lower speed than the prime mover. The other industrial machines where gear speed reducers are required are: rolling mills used in steel industries, hoists used for different applications, integrated high torque electrical motors, etc. In all these applications, the physical look of the reducer can be different but the basic elements such as gears, shafts, bearings, etc.,used in it, remain the same. Because of this fact, more attention is paid towards the design of these basic elements.

The implementation points towards the idea of designing a Gear speed reducer, along with its different components, in an interactive fashion in front of the terminal. This idea came out from the fact that the mechanical systems, when designed on the desk, do take many more man hours. More time is spent on arithmatic calculations and referring the standard codes. These type of problems are routine problems for which an algorithmic procedure can be developed. Material properties, form factor for gear tooth, etc., which are available in the form of a table or

chart, are reffered many a times in the design process. Designing with the help of a computer can be a good solution for such problems. These are best suited for work which can be put in the form of an algorithm. Standard data, which is necessary for an economical and practical design, can be stored and accessed whenever required. These standards are provided to ensure uniformity in design. However, a typical design can have nonstandard elements incorporated in it.

Interactive design of each component and then their integration to form a mechanical system, viz. a gear speed reducer, has been attempted in this work.

1.2 LITERATURE SURVEY

Various methods have been suggested for the design of gears, shafts, bearings, etc., in different texts [1,2,3,4,5,8] and papers [10,12,14]. This design is dependent on the standard data used for it. Every country has its own standardization scheme. The design procedure is also different for each standard. Standard data for mechanical components can be referred from ASME (American Society of Mechanical Engineers) code, BS (British Standards) code, ISI (Indian Standards Institution) code, etc. Design data [2] which contains the ISI code and relevent information about the design of machine elements, is used as a reference text in this work. Cockerham and Waite [10] have used British standards in their implementation for designing a spur and bevel gear. Interactive programs have been developed for the design process. Prayoonrat and Walton [11], suggested a practical

approach for the design of a gear train. A scheme for optimizing overall size was presented which can take into account different criteria of optimization such as : optimization with respect to minimum overall center distance, minimum gear volume, maximum contact ratio, etc. A direct search method has been used to remove unacceptable solutions and to locate quickly the region where the optimum design lies. The final dimensions of the gears necessary for the manufacturing purpose have been calculated. Mabie and Mitchner [9], have given relevent mathematical theory for determining the Lewis form factor Y and AGMA geometry factor J for a spur gear tooth. Geometry of root fillet of gear tooth, consistent with the locus of hob tooth tip, has been described. Tsay, Chung-Biau [12], has given an algorithm for determining stress distributions in the involute tooth of a helical gear. A mathematical model and computer simulation of the meshing and bearing contact has been presented. Sensitivity of gears to the errors of manufacturing and assembly has been investigated. Carroll and Johnson [13], presented a scheme for the design of a compact spur gear set with minimization of pinion diameter. Optimization of the weight of gear box has been done with respect to gear dimensions. Kinzel and Walliser [17], in their work, outlined how to structure interactive design programs typically applicable to mechanical element design. General structure and descipline to be maintained in the program development has been pointed out.

1.3 SCOPE OF THE PROBLEM

In the present work three basic types of gears namely

spur, helical (herringbone) and bevel, have been included as the basic torque transmitting elements. Full depth involute type tooth form with twenty degrees pressure angle has been considred. Antifriction bearings are used as shaft support elements. Different types of bearings, depending upon the requirement, except thrust and needle, have been included in the system. The system is designed in such a way that it gives sufficient flexibility in taking decisions at critical stages of design. Effect of the change in the value of a parameter for example; and , can be seen immediately on the screen. Hence, the present system can be used by designers as a major tool in their design process.

The problem can be stated as follows;

"Given the power transmitted by the system, input or output speed, number of stages, expected life of the system and type of gears used in it, to design gears, shafts, bearings, keys and sleeves of the gear speed reducer for a specified working condition."

This problem has been broken into separate modules. Every part has been designed separately in a hierarchial design process. While integrating them, dimensional checks have been incorporated viz. possible interference of two gears in adjacent stages can be checked, etc.

1.4 ORGANIZATION OF THE THESIS

Chapter 2 describes in detail the design strategy of different components used in the system. Algorithms and diagrams,

describing the development of pertinent equation, are presented wherever necessary.

Chapter 3 deals with the implementation part of the problem. Detailed information about various data structures used in the program are pointed out. Block diagrams illustrating the data structure are presented. A methodology to incorporate dynamic and static data base management in the system is given in detail. Salient features of the packages used in the system are given.

Chapter 4 gives an idea about the performance of the system. The results obtained from a trial run of the system are presented. Graphic output is shown with the help of photographs.

In chapter 5, directions for further work and limitations in the system are pointed out. This chapter concludes the thesis report.

Appendix 1 includes the user manual for the system. Brief description of prompts appearing on the screen are mentioned.

CHAPTER 2

INTEGRATED SYSTEM DESIGN

2.1 INTRODUCTION

A gear speed reducer contains different elements which are required to be designed separately. Before designing a particular element it is made sure that all the predecessors have been designed completely; which means that a proper sequence of elements has to be maintained during design. Flow chart which depicts this, is shown in fig. 2.1. Basic elements which are incorporated in the system are as follows.

1. gears
2. shafts
3. bearings
4. keys
5. sleeves

This chapter describes in detail the design procedures of the machine elements mentioned above. Algorithms and flow charts are given wherever necessary.

2.2 DESIGN OF GEARS

Following are the type of gears which have been considered in the system as the basic gear elements;

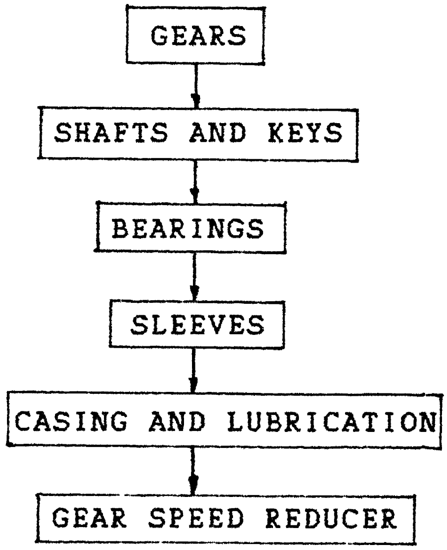


Fig. 2.1 Hierarchy of design of elements.

1. Spur

2. Helical (Herringbone)

3. Straight Bevel

Bevel gears are included in the system to facilitate change in the direction of power transmission through right angles. They can be put anywhere in multistage gear train. Placing them in the middle of the train makes the design more complex. Hence, they are placed either at the beginning or end always. Furthermore, in multistage design, the final stage transmits the maximum torque. Therefore, the best position at which bevel gears can be placed is at the first stage. It is subjected to the minimum torque and hence its size becomes smaller.

2.2.1 Material Properties

Various mechanical properties such as surface compressive stress, bending strength, etc., are required for which experimentally verified ralations may be used. Stress in square brackets denote design stress. Following are the relations by which they can be approximated.

Design Surface Compressive Stress [σ_c] [2]

The surface compressive strength of the material is proportional to the surface hardness of the material and is given by

$$\sigma_c = \text{BHN CB} \quad \text{kg/sq.cm}$$

$$= \text{HRC CR} \quad \text{kg/sq.cm} \qquad \text{... (1)}$$

where CB and CR are constants [2] whose value depends on the material and its heat treatment (table 2.1).

To take into account the loading conditions, factor k_{cl} is introduced. It is defined [2] as follows

k_{cl} = 0.585 if BHN > 350 and N ≥ 25 10^7 ... (2)

= 1.0 if BHN ≤ 350 and N ≥ 10^7 ... (3)

= $(10^7 / N)^{1/6}$ for Cast Iron and all other cases .(4)

where N is the total no of fatigue cycles in the life time.

The design surface compressive strength, thus, becomes

$[\sigma_c]$ = BHN CB k_{cl} kg/sq.cm ... (5)

Endurance Stress in Bending σ_{-1}

Following are the relations [2] from which endurance limit stress in bending, close to experimental value, can be calculated

Forged steels $\sigma_{-1} = 0.25 (\sigma_u + \sigma_y) + 500$ ... (6)

Cast steels $\sigma_{-1} = 0.22 (\sigma_u + \sigma_y) + 500$ ... (7)

Alloy steels $\sigma_{-1} = 0.35 \sigma_u + 1200$ ... (8)

Cast iron $\sigma_{-1} = 0.45 \sigma_u$ ... (9)

Design Bending Stress $[\sigma_b]$ [2]

A gear tooth is always subjected to cyclic bending stresses. The $[\sigma_b]$ is calculated [2] from bending endurance limit stress with suitable factor of safety n (table 2.2) introduced to take care of the type of material and its heat treatment.

$$[\sigma_b] = k_{b1}\, \sigma_{-1} / (n\, k\ )\ \text{kg/sq.cm} \qquad \ldots (10)$$

for gears with two directions of rotation

$$[\sigma_b] = 1.4\, k_{b1}\, \sigma_{-1} / (n\, k\ )\ \text{kg/sq.cm} \qquad \ldots (11)$$

for gears with only one direction of rotation

where k_{c1} is the life factor for bending stress which is calculated as follows

$$k_{b1} = 0.7 \text{ if } N \geq 25\ 10^7 \text{ and BHN} > 350 \qquad \ldots (12)$$

$$= 1.0 \text{ if } N \leq 10^7 \qquad \ldots (13)$$

$$= (10^7 / N)^{1/9} \text{ for C.I. and all other cases} \qquad \ldots (14)$$

Equivalent Young's Modulus E

Equivalent Young's Modulus for two materials in contact can be given [2] by

$$E = 2\, E_1\, E_2 / (E_1 + E_2) \qquad \ldots (15)$$

Where E_1 and E_2 are Young's Moduli of the materials concerned.

Criterion of Design

Gears are designed either with respect to tooth bending strength or surface compressive strength [1]. If both the gears are made of steel, surface compressive stress governs the design. But if one of them is made of cast iron, design should be based on bending strength of tooth, since cast iron is weaker in bending.

STRESS CONCENTRATION FACTORS

(1) Load concentration factor k

During run conditions, the load on the tooth is not uniformly distributed over the line of contact because teeth get deformed and change their mutual positions. This is also due to inaccuracies in machining of wheels, shafts and housing and assembly errors. If q_{max} is the maximum unit load due to uneven distribution along the line of contact then load concentration factor k is gien by

$$k = q_{max} / q \qquad \ldots (16)$$

where q is average unit load along the line of contact.

It depends on the following

1. Location of gears relative to bearings
2. Rigidity of the shaft.
3. The factor .
4. Rigidity of the teeth of gear and pinion.
5. Rigidity of gear and pinions.

table 2.3 contains information about the value of k for various conditions.

(2) Dynamic load factor k_d

Inaccuracies in contact due to inaccuracies of tools and tooth cutting machines cause jerky operation of toothed gears. This causes angular acceleration and deacceleration of the gears even though angular velocity of driving wheel is constant. Shock is also possible in accurately machined gears because of the teeth deformation under load. The pitches of a pair of gears change and on the gear it increases and on pinion it decreases. This shock causes additional load on tooth. This depends on the degree of accuracy, stiffness of teeth, surface hardness and

pitch line velocity.

If F_i is the increment in load due to dynamic action, the total load on the tooth,

$$F_d = F_n + F_i \quad \ldots (17)$$

$$= F_n [1 + F_i / F_n] \quad \ldots (18)$$

$$= k_d \;\; F_n \quad \ldots (19)$$

table 2.4 contains the value of k_d.

(3) Stress concentration factor k_σ

The stress concentration factor is introduced to take care of stress intensity at the root fillet. This factor depends on the type of material and its heat treatment. The values are given in table 2.5 for different materials.

Check for plastic deformation of the tooth

(1) Under σ_c

Instanteneous overloads on the tooth may cause plastic deformation of the tooth surface even though such load will not affect the surface endurance. The surface strength of material for instantaneous loads are

$$[\sigma_c]_{max} = 3.1 \; \sigma_y \quad \text{if BHN} \leq 350 \text{ for steel} \quad \ldots (20)$$

$$= 420 \text{ HRC if BHN} > 350 \text{ for steel} \quad \ldots (21)$$

$$= 1.8 \; \sigma_u \quad \text{for Cast Iron} \quad \ldots (22)$$

If $[M_t]_{max}$ is the maximum instantaneous torque on the pinion and $\sigma_{c\ max}$ is the surface stress induced due to above torque then

$$\sigma_{c\ max} = \sigma_c \sqrt{((M_t)_{max} / M_t)} \quad \ldots (23)$$

$$\sigma_{c\ max} < [\sigma_c]_{max} \quad \ldots (24)$$

(2) Under σ_b

The maximum bending stress induced due to $(M_t)_{max}$ is given by

$\sigma_{b\ max} = \sigma_b\ (M_t)_{max} / M_t$ ... (25)

$\sigma_{b\ max} \le [\sigma_b]_{max}$ ... (26)

where the value of $[\sigma_b]_{max}$ is given as follows

$[\sigma_b]_{max} = 0.8\ \sigma_y$ if BHN $\le$ 350 and no heat treatment (27)

$= 0.36\ \sigma_u / k_\sigma$ if heat treated and BHN $\ge$ 350 (28)

$= 0.6\ \sigma_u$ ... (29)

2.2.2 Design of spur gears

The methodology of design of gears is given in the form of an algorithm where the equations used are shown in the parantheses.

Algorithm

step 1. Calculate a. equivalent Young's Modulus and b. Design Bending, Endurance and Surface compressive stress from the given material data (1 - 29).

step 2. Accept the number of teeth in pinion from the user.

step 3. Decide upon the criterion of design

step 4. if gears are to be designed with respect to bending stress

step 4.1. calculate minimum module necessary to keep bending stress within limit (31)

else

step 4.1. calculate minimum center distance necessary to keep surface stress within limit (30)

step 4.2. calculate module corresponding to this center distance (32)

step 5. Select standard module higher in magnitude than obtained module (table 2.6).

step 6. Calculate face width

step 7. Check for

plastic deformation of tooth under static load

induced bending stress (34)

induced surface compressive stress (33)

print the status of all stresses; dimensions of gear and pinion

step 8. Check with the user whether the design is satisfactory or not.

step 9. if design is not satisfactory ask to change the values of the following variables

number of teeth in pinion

ratio b/a

ratio b/m

Material properties

set new values

go to step 3

else

step 10. stop

Coefficients CB and CR [2]

Material Hardness	Heat Treatment	Surface CB or	Coeff. CR
Carbon steels and alloy steels any type	Normalized or Hardened or Temptered	BHN ≤ 350	CB = 25
High strength alloy Nickle Chromium steels	Case Hardened	BHN 540 to 540	CR =310
Alloy steels	"	"	CR =280
Carbon and Manganese Steels	"	"	CR =220
Alloy steels; carbon steels	Hardened and Tempered	BHN 375 to 540	CR =265
"	Surface Hardened	"	CR =230
Cast Iron, Grade 20, 25	-	BHN 170 to 200	CB = 20
Cast Iron, Grade 30, 35	-	BHN 200 to 260	CB = 23

table 2.1

Factor of Safety n [2]

Material	Mode of Manufacture	Heat treatment	n
steel, Cast Iron	Cast	No Heat treatment Tempered or Normalized	2.5 2.0
Steel	Cast or forged Forged Forged	Case Hardened Surface Hardened Normalized	2.0 2.5 2.0

table 2.2

Load concentration factor k

= b/a	Bearing very close to gears (symm.)	Unsymmetrical rigid l/d < 3	Unsymmetrical less rigid l/d < 3	Cantilever
0.2	1.0	1.0	1.05	1.15
0.4	1.0	1.04	1.10	1.22
0.6	1.03	1.08	1.16	1.32
0.8	1.06	1.13	1.22	1.45
1.0	1.10	1.18	1.29	-
1.2	1.14	1.23	1.36	-
1.4	1.19	1.29	1.45	-
1.6	1.25	1.35	1.55	-

table 2.3

Dynamic load factor k_d

IS quality	BHN	Pitch line velocity, m/s < 1	1 - 3	3 - 8	8 - 12
5	≤350	-	-	1.2	1.3
	>350	-	-	1.2	1.3
6	≤350	-	1.25	1.45	1.55
	>350	-	1.2	1.30	1.40
7	≤350	1.00	1.35	1.55	-
	>350	1.00	1.30	1.40	-
8	≤350	1.10	1.45	-	-
	>350	1.10	1.40	-	-

table 2.4

Stress concentration factor k_σ

Material and treatment	k_σ
steel, normalized, surface hardened	1.5
steel, case hardened	1.2
cast iron	1.2

table 2.5

IS: 2535 - 1963 Recommended Series of Modules mm [2]

Choice 1	Choice 2	Choice 3
1.0		
1.25	1.125	
1.5	1.375	
2.0	1.75	
2.5	2.25	
3.0	2.75	3.25
4.0	3.5	3.75
5.0	4.5	
6.0	5.5	
8.0	7.0	6.5
10.0	9.0	
12.0	11.0	
16.0	14.0	
20.0	18.0	

table 2.6

Form Factor, y for α = 20 [2]

Z or Z_v	y
12	0.308
14	0.330
16	0.355
18	0.377
20	0.389
22	0.402
24	0.414
26	0.427
28	0.434
30	0.440
35	0.452
40	0.465
45	0.471
50	0.477
60	0.490
80	0.499
100	0.505
150	0.515
300	0.521
RACK	0.550

table 2.7

Design Formulae [2]

$$a \geq (i + 1) \sqrt[3]{\left[\frac{0.74}{[\sigma_c]}\right]^2 \frac{E\ [M_t]}{i\ \psi}} \qquad \ldots (30)$$

$$m \geq 1.26 \sqrt[3]{\frac{[M_t]}{y\ [\sigma_b]\ Z_1}} \qquad \ldots (31)$$

$$m = 2\ a\ /\ (Z_1 + Z_2) \qquad \ldots (32)$$

Checking induced stresses [2]

$$\sigma_c = 0.74 \frac{(i + 1)}{a} \sqrt{\frac{(i + 1)\ E\ [M_t]}{i\ b}} \leq [\sigma_c] \qquad \ldots (33)$$

$$\sigma_b = \frac{(i + 1)\ [M_t]}{a\ m\ b\ y} \leq [\sigma_b] \qquad \ldots (34)$$

2.2.3 Design of Helical and Herringbone gears [8]

Algorithm

step 1. Calculate a. Equivalent Young's Modulus and b. Design Bending, Endurance and Surface compressive stress from the given material data (1 - 29).

step 2. Accept the no. of teeth in pinion from user.

step 3. Accept direction of helix in pinion.

step 4. Decide upon the criterion of design

step 5. if gears are to be designed with respect to bending stress

step 5.1. calculate minimum normal module necessary to keep bending stress within limit (37)

else

step 5.1. calculate minimum center distance

necessary to keep surface stress within limit (36)

step 5.2. calculate normal module corresponding to this center distance (37)

step 6. Select standard normal module higher in magnitude than obtained normal module (table 2.6).

step 7. Calculate face width

step 8. Check for

plastic deformation of tooth under static load

induced bending stress (39 or 40)

induced surface compressive stress (38)

print the status of all stresses; dimensions of gear and pinion

step 9. Check with the user whether the design is satisfactory or not.

step 10. if design is not satisfactory

ask to change the values of the following variables

number of teeth in pinion

ratio b/a

ratio b/m

Material properties

set new values

go to step 4

else

step 11. stop

Design Formulae [2]

$$a \geq (i + 1) \sqrt[3]{\left[\frac{0.70}{[\sigma_c]}\right]^2 \frac{E\ [M_t]}{i\ \Psi}} \qquad \ldots (35)$$

$$m_n \geq 1.15 \cos\beta \sqrt[3]{\frac{[M_t]}{y_v\ [\sigma_b]\ Z_1\ \Psi_m}} \qquad \ldots (36)$$

$$m_n = 2\ a \cos\beta\ /\ (Z_1 + Z_2) \qquad \ldots (37)$$

Checking induced stresses [2]

$$\sigma_c = 0.70\ \frac{(i + 1)}{a} \sqrt{\frac{(i + 1)\ E\ [M_t]}{i\ b}} \leq [\sigma_c] \qquad \ldots (38)$$

For the helical gears

$$\sigma_b = 0.7\ \frac{(i + 1)\ [M_t]}{a\ m_n\ b\ y_v} \leq [\sigma_b] \qquad \ldots (39)$$

For the herringbone gears

$$\sigma_b = 0.85\ \frac{(i + 1)\ [M_t]}{a\ m_n\ b\ y_v} \leq [\sigma_b] \qquad \ldots (40)$$

2.2.4 Design of bevel gears

Algorithm

step 1. Calculate a. Equivalent Young's Modulus and b. Design Bending, Endurance and Surface compressive stress from the given material data (1 - 29).

step 2. Accept no. of teeth in pinion from the user.

step 3. Decide upon the criterion of design

step 4. if gears are to be designed with respect to bending stress

step 4.1. calculate minimum average module

necessary to keep bending stress within limit (42)

step 4.2. calculate transverse module corresponding to this average module (33)

else

step 4.1. calculate minimum cone distance necessary to keep surface stress within limit (41)

step 4.2. calculate transverse module corresponding to this cone distance (44)

step 5. Select standard module higher in magnitude than obtained module (table 2.6).

step 6. Calculate face width

step 7. Check for

plastic deformation of tooth under static load

induced bending stress (46)

induced surface compressive stress (45)

print the status of all stresses; dimensions of gear and pinion

step 8. Ask whether the design is satisfactory or not

step 9. if design is not satisfactory

ask to change the values of the following

number of teeth in pinion

ratio b/R

ratio R/m

Material properties

set new values

go to step 3

else

step 10. stop

Design Formulae [2]

$$R \geq \sqrt{(i^2 + 1)}\ \sqrt[3]{\left[\frac{0.72}{(\Psi_y - 0.5)\ [\sigma_c]}\right]^2 \frac{E\ [M_t]}{i}} \qquad \ldots (41)$$

$$m_{av} \geq 1.28\ \sqrt[3]{\frac{[M_t]}{y_v\ [\sigma_b]\ Z_1\ \Psi_m}} \qquad \ldots (42)$$

$$m_t = m_{av} + b \sin\delta\ /\ Z \qquad \ldots (43)$$

$$m_t = 2\ R\ /\sqrt{(Z_1^2 + Z_2^2)} \qquad \ldots (44)$$

Checking induced stresses [2]

$$\sigma_c = \frac{0.72}{(R - 0.5b)} \sqrt{\frac{\sqrt{(i^2+1)^3}\ E\ [M_t]}{i\ b}} \leq [\sigma_c] \qquad \ldots (45)$$

$$\sigma_b = \frac{R \sqrt{(i^2 + 1)}\ [M_t]}{b\ m\ y_v\ (R - 0.5b)^2\ \cos\alpha} \leq [\sigma_b] \qquad \ldots (46)$$

2.2.5 Generation of Involute tooth and root fillet

The gears considered in this system are of involute type. The process of generating the involute flank profile on an external spur gear tooth may be visualized and treated as the ...ing of a rack with a straight flank hob profile on the pitc surface of the gear blank. To generate the involute flank profile of the gear tooth, following procedure is adopted.

An involute of a given curve C is a curve I such that C is the locus of the centers of curvature of I. In gearing theory, the curve C is always a circle, called base circle, and an involute is visualized as being created by unwrapping a string from the base circle (fig. 2.2).

Tangent and Normal coordinates of an involute

Consider a point P on an involute curve. The tangent at P is along the x axis and the normal at P is along the y axis. Coordinates of other point Q can be found out [3] with the help of following equations

$$x = r\,[\sin\phi \sin\theta + \cos\phi\,(\theta \sin\theta + \cos\theta - 1)] \qquad \ldots (47)$$

$$y = r\,[\sin\phi\,(1 - \cos\theta) + \cos\phi\,(\sin\theta - \theta \cos\theta)] \qquad \ldots (48)$$

where r = radius of point P

ϕ = pressure angle at point P

θ = angle between radii drawn to two points on the base circle, one on the normal through P and the other on the normal through Q.

θ and x are both positive when Q is above P and both negative when Q is below P. y is always positive and is measured toward the gear tooth center line. At P, x and y are both positive.

Both r and ϕ are fixed for a given selection of P, so that additional points Q on the involute can be calculated by varying θ only. The point P is usually taken to be the pitch point , in which case r is the pitch radius of the gear and ϕ is its pressure angle. If R is the radius of point Q and its pressure angle is ϕ, the theta value for point Q is equal to the difference between $\tan\phi_1$ and $\tan\phi$ when θ is measured in radians. The theta can be found out as

$$\theta = \sqrt{\{(R / (r \cos\phi))^2 - 1\}} - \tan\phi \qquad \ldots (49)$$

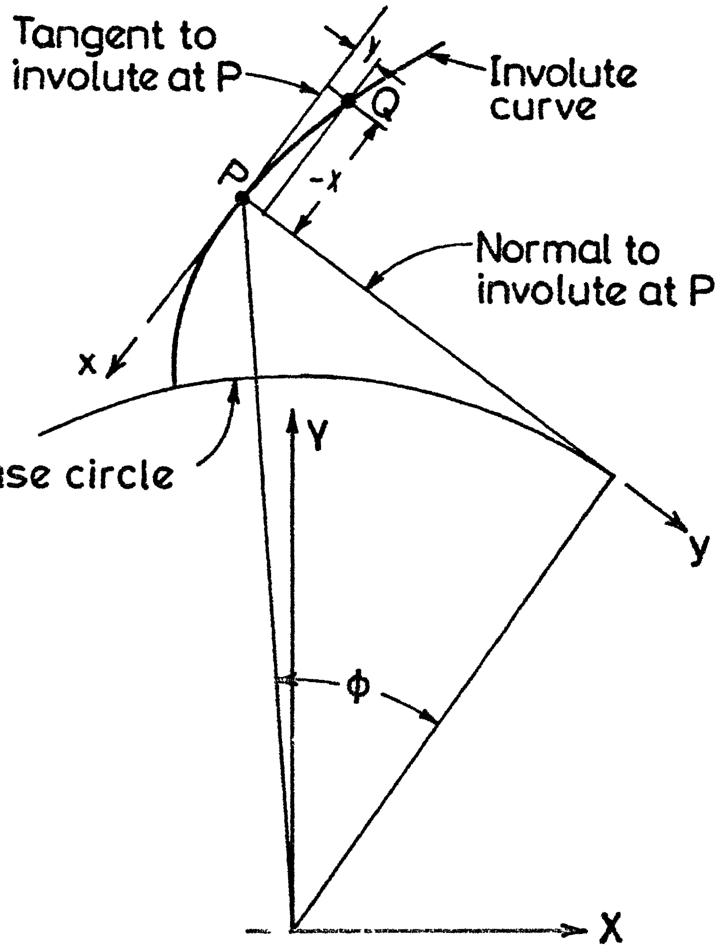


Fig. 2.2 Tangent and normal coordinates of involute.

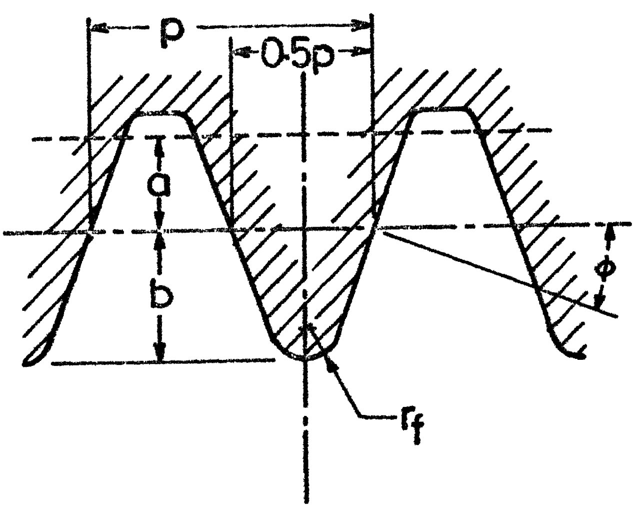


Fig. 2.3 Hob profile

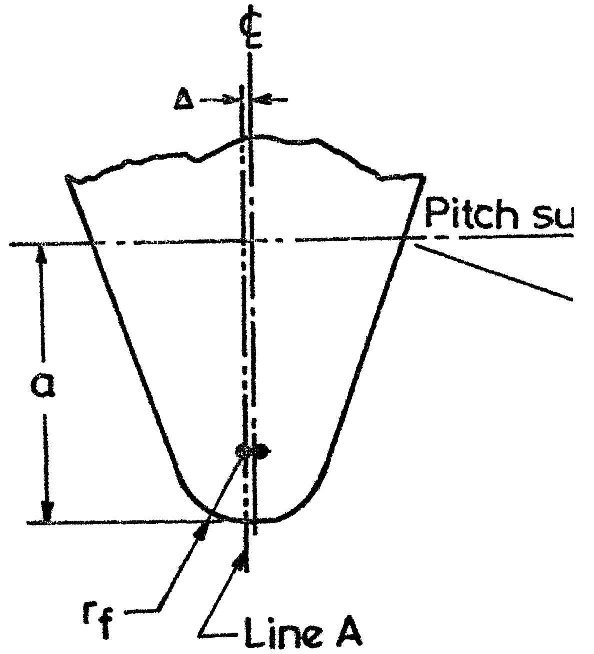


Fig. 2.4 Enlarged hob-tooth

Involute curve exists only between base circle and addendum circle. Hence the points lying on the curve can be calculated with the help of formulae given above.

The root fillet [9] is generated as follows. Consider the hob profile shown in Fig. 2.3 with an enlarged view of the hob tip as shown in Fig. 2.4. The distance Δ, which is the half width of the hob tip land is described by

$$\Delta = (\pi / (4 P)) - (b - r_f)\tan\emptyset - r_f / \cos\emptyset \qquad \ldots (50)$$

For the condition when $\Delta = 0$, the hob tip radius is given by

$$r_f = \{(\pi \cos\emptyset / (4 P)) - b \sin\emptyset \} / (1 - \sin\emptyset) \qquad \ldots (51)$$

Line A of Fig. 2.4 is offset from the centerline of hob tooth by the distance Δ and this line passes through the center of the hob tip radius r_f. When the hob is engaged with the blank as shown in Fig. 2.5 and positioned such that the centerline of a blank tooth lies on the vertical axis, generation of the blank tooth begins when line A of the hob is collinear with line F at the blank. Line F makes an angle η with the centerline of the space where η is described by

$$\eta = \Delta / R \qquad \ldots (53)$$

where R is pitch circle radius

If the pitch surface of the hob is then rolled about the pitch circle of the blank to the right, the centerline of the hob

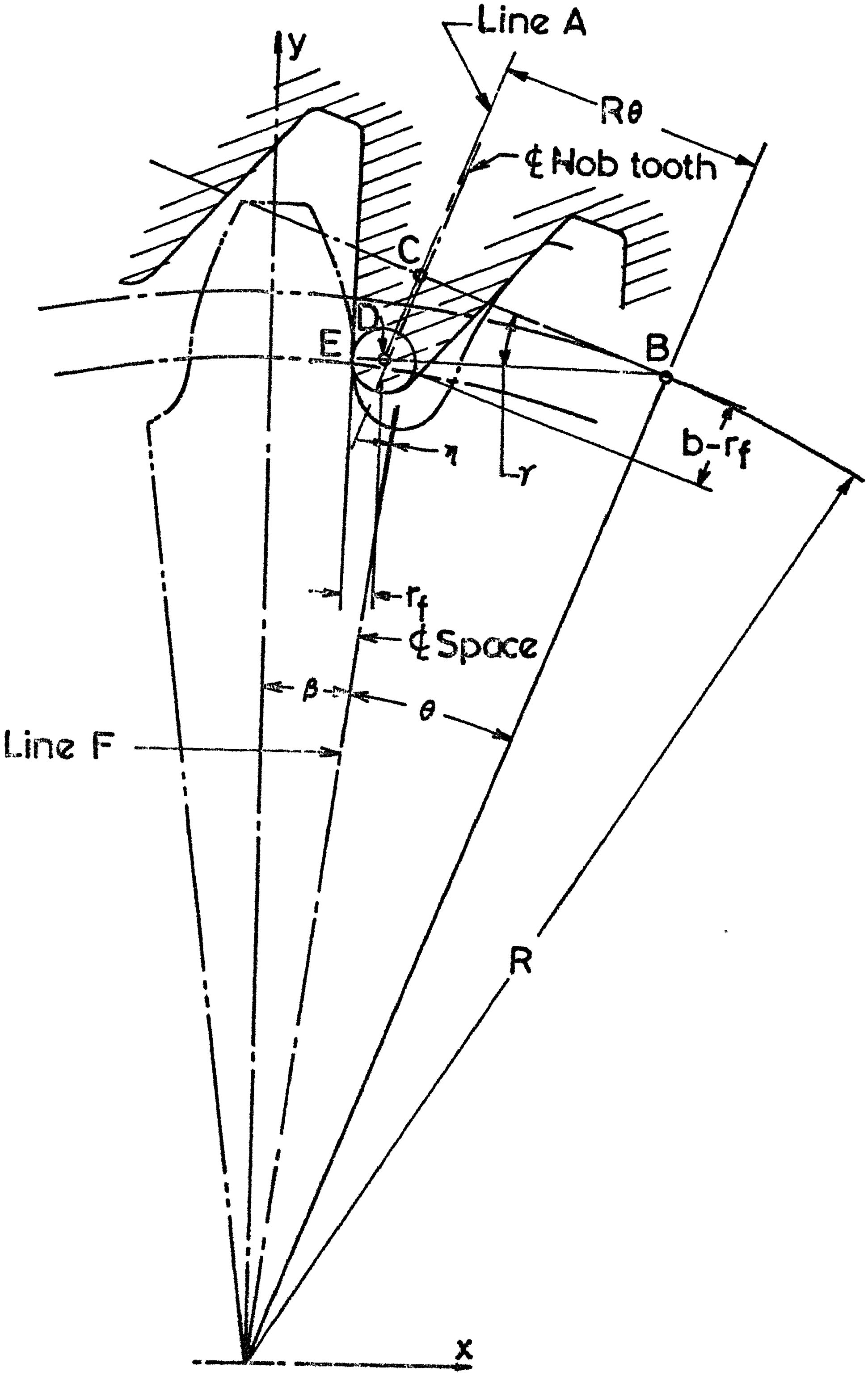


Fig. 2.5 Engagement of hob and the gear blank.

tooth will become collinear with the centerline of the tooth space. The angular positions of point of tangency, point B, of the two pitch surfaces can be described by the angle θ which is measured from line F. As the hob is rolled on the blank, point B becomes the instantaneous center of rotation. The path of point E will describe the envelope of the path of points on the hob tooth in the generation of the root profile, provided that Γ is greater than the cutting pressure angle Ø.

Consider Fig 2.6 where only the envelope of the path of points on the hob tooth are of interest. Point C will have an involute path as θ is increased. The angle BCD is a fixed right angle as the blank tooth root is generated. Point D will have a trochoidal path described parametrically as

$$x_D = -R\,\theta\cos(\beta + \theta) + (R - b + r_f)\sin(\beta + \theta) \quad \ldots (53)$$

$$y_D = R\,\theta\sin(\beta + \theta) + (R - b + r_f)\cos(\beta + \theta) \quad \ldots (54)$$

Since DE must be collinear with BD as the generation proceeds, the parametric coordinates of Point E, which lie on the trochoid are

$$x_E = (R - b + r_f)\sin(\beta + \theta) - R\,\theta\cos(\beta + \theta)$$
$$- r_f/[\{(b-r_f)\sin(\beta+\theta) + R\theta\cos(\beta+\theta)\}\surd((b-r_f)^2+R^2\theta^2)] \quad \ldots (55)$$

$$y_E = (R - b + r_f)\cos(\beta + \theta) + R\,\theta\sin(\beta + \theta)$$
$$- r_f/[\{(r_f-b)\cos(\beta+\theta) + R\theta\sin(\beta+\theta)\}\surd((b-r_f)^2+R^2\theta^2)] \quad \ldots (56)$$

where N is total no of teeth and β is given by

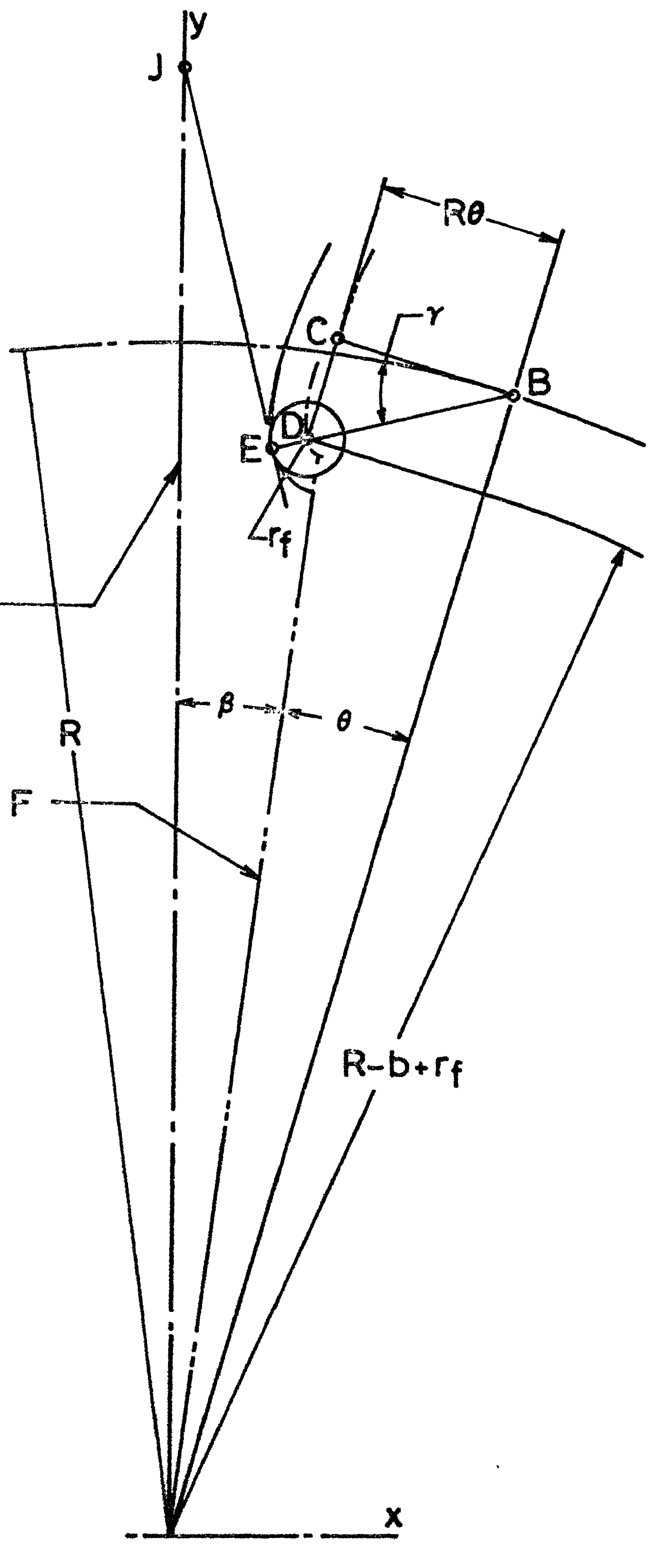


Fig. 2.6 Generation of root trochoid.

$$\beta = (\pi / N) - \eta \qquad \ldots (57)$$

Hence, points lying on the fillet can be found out for θ varying between 0 to a value for which polar distance of (x_E, y_E) becomes equal to the radius at which tooth curve ends.

2.3 Selection of bearings

Antifriction bearings can be selected for a particular application if the status of load at the point of application and the dynamic capacity required is knowm. The following are the equations necessary to compute dynamic capacity required

The equivalent load P_{eq} [2] is determined by

$$P_{eq} = (X P_r + Y P_a) S \qquad \ldots (58)$$

To find the rating life [2] of bearing at a given probability, the following equation is used

$$L / L_{10} = [\ln(1/p) / (\ln(1/p_{10})]^{1/b} \qquad \ldots (59)$$

where b = 1.17 for a median life of $5L_{10}$ [2]

b = 1.34 for deep groove ball bearing

$$C = (L / L_{10})^{1/k} P_{eq} \qquad \ldots (60)$$

where k = 3 for ball bearing [1]

k = 10/3 for roller bearing

Algorithm

step 1. find out e (P_a / P_r) from the given force components

step 2. select the bearing type to be used

step 3. if selected bearing is roller bearing and e is non zero

go to step 2

step 4. select X and Y factors (table 2.8)

step 5. find out equivalent load on the bearing (58)

step 6. calculate rating life at 90% probability of survival (59)

step 7. compute dynamic capacity needed (60)

step 8. find out a bearing from the standard set which has inner diameter greater than or equal to the shaft diameter

step 9. print obtained and standard dynamic capacites and % difference between them

step 10. Check with the user whether the selection should be accepted or not

step 11. if not accepted

go to step 2

else

step 12. stop

2.4 Design of shafts [1]

Following factors are included for different loading and working conditions.

(1) Column action factor α

The column action factor is introduced as a factor of safety against buckling. This is given as follows

$\alpha = 1$ if P_a is tensile

$$\alpha = 1 \;/\; (1 - 0.0044\,(l/r)) \text{ for } l/r < 150 \qquad \ldots (61)$$

(2) Factors K_t and K_b

The values of above factors are given in table 2.9

These factors are incorporated to calculate the equivalent twisting moment for the design of shaft.

For designing the shaft, the method proposed by ASME has been used. The development of the equation is based on the maximum shear stress theory of failure. The approach gives conservative results. Since the inner diameter of the bearing cannot be changed, the of shaft diameter will be rounded off to inner diameter of available bearing.

$$d_1^{\,3} = \frac{16}{\pi\,[\tau]} \sqrt{[K_b\, M_b + \alpha\,(P_a\, d \,/\, 8)]^2 + (K_t\, M_t)^2} \qquad \ldots (62)$$

The code defines the value of permissible shear stress as

$$\tau = \min\,[\,0.3\sigma_y\,,\ 0.18\sigma_u] \qquad \ldots (63)$$

Accordingly, τ should be reduced further by 25% if stress concentration, possibly due to shoulder fillet or keyway, is present.

In this system shafts are stepped and hence the design shear stress of the material is given by

$$[\tau] = 0.75\,\tau \qquad \ldots (64)$$

The diameter obtained above can be checked further for endurance stress to take in to consideration the fatigue of the material due to loading.

Selection of Factors X and Y for bearings

Type of bearing	Series	F_a/C_o	$F_a/F_r \leq e$ X	$F_a/F_r \leq e$ Y	$F_a/F_r \geq e$ X	$F_a/F_r \geq e$ Y	e
Deep groove Ball Bearing	60 62 63 64	0.025	1	0	0.56	2	0.22
		0.04	1	0	0.56	1.8	0.24
		0.07	1	0	0.56	1.6	0.27
		0.13	1	0	0.56	1.4	0.31
		0.25	1	0	0.56	1.2	0.37
		0.50	1	0	0.56	1.0	0.44
Angular Contact Ball bearing	72 73	-	1	0	0.35	0.57	1.14
		-	1	0	0.35	0.57	1.14
		-	1	0.55	0.57	0.93	1.14
		-	1	0.73	0.62	1.17	0.86
Taper Roller Bearing	322 323	-	1	0	0.4	1.6	0.37
		-	1	0	0.4	1.45	0.41
		-	1	0	0.4	1.35	0.44
Self Aligning Ball bearing	22 23	-	1	1.3	0.65	2.0	0.5
		-	1	1.7	0.65	2.6	0.37
		-	1	2.0	0.65	3.1	0.31
		-	1	2.3	0.65	3.5	0.28
		-	1	2.4	0.65	3.8	0.26
		-	1	2.3	0.65	3.5	0.28
Spherical Roller Bearing	222	-	1	2.1	0.67	3.1	0.32
		-	1	2.5	0.67	3.7	0.27
		-	1	2.9	0.67	4.4	0.23
		-	1	2.6	0.67	3.9	0.26

table 2.8

Selection of factor K_t and K_b

Type	K_b	K_t
Gradual Loading	1.5	1.0
Minor shock loads	1.5-2.0	1.0-1.5
Heavy Shock loads	2.0-3.0	1.5-3.0

table 2.9

material due to loading.

$$d_2^3 = \frac{32 \sqrt{(M_b^2 + M_t^2)}}{\pi [\sigma_{-1}]} \qquad \text{... (65)}$$

The diameter of shaft, d, can now be given as

$$d = \max[d_1 , d_2] \qquad \text{... (66)}$$

2.5 Design of keys

Keys are the machine elements which are necessary to prevent relative angular motion between two concentric and overlapping bodies. These can either be designed with respect to the allowable shear stress of key material and allowable compressive stress of key and shaft material or selected from the standard set and further checked for induced stresses.

2.5.1 Designing with respect to allowable stresses

Keys can fail either in shear or compression. If the length of key is known, the width can be calculated by considering failure on the plane parallel to the direction of force. The thickness of the key can be determined if allowable stresses in compression are considered.

$$w = 2 [M_t] / (d\ l\ [\tau]) \qquad \text{... (67)}$$

$$t_1 = 4 [M_t] / (d\ l\ [\sigma_c]_{key}) \qquad \text{... (68)}$$

$$t_2 = 4 [M_t] / (d\ l\ [\sigma_c]_{shaft}) \qquad \text{... (69)}$$

and the thickness

$$t = \max[t_1 , t_2] \qquad \text{... (70)}$$

2.5.2 Selecting and then checking for induced stresses

Standard dimensional data-base [2] of rectangular parallel key can be searched for the given diameter of shaft as the key value. The length of key is usually taken equal to the gear hub width. Width and thickness of the rectangular key can thus be found out.

The induced strress τ is given by

$$\tau = 2\ [M_t] / (d\ l\ w) \quad kg/sq.cm \qquad \ldots (71)$$

and

$$\tau \leq [\tau] \qquad \ldots (72)$$

approximate induced compressive stress in key material

$$(\sigma_c)_{key} = 4\ M_t / (d\ l\ t) \quad kg/sq.cm \qquad \ldots (73)$$

and

$$(\sigma_c)_{key} \leq [\sigma_c]_{key} \qquad \ldots (74)$$

approximate induced compressive stress in shaft material

$$(\sigma_c)_{shaft} = 4\ M_t / (d\ l\ t) \quad kg/sq.cm \qquad \ldots (75)$$

and

$$(\sigma_c)_{shaft} \leq [\sigma_c]_{shaft} \qquad \ldots (76)$$

should be satisfied to avoid failure.

2.6 Design of sleeve

Sleeves are used in the reducer to keep the gear elements separate from each other and to maintain a safe distance from the wall. These are usually made of steel and are in the form of a hollow cylinder. The inner diameter of this cylinder is equal to the diameter of shaft on which it is slept. Since this member is not subjected to twisting moment and tensile load, the outer

diameter is calculated such that the induced compressive stress is less than the compressive strength of the material.

The thickness t to be designed is found out as follows

$$t > P_a / (\pi D_{inner} [\sigma_c]) \text{ kg/sq.cm} \quad \ldots (77)$$

if t is assumed, the induced stress can be checked as follows

$$\sigma_c = P_a /(\pi D_{inner} t) \text{ kg/sq.cm} \quad \ldots (78)$$

$$\sigma_c < [\sigma_c] \quad \ldots (79)$$

The length of the sleeve is equal to the distance between the bearing inner race and side face of gear hub.

2.7 General approach of design

A sequential approach for designing the integrated system is as follows. Information about life expectency, working conditions, probability of survival of each bearing, etc., are required before going for design. At first, the gears required in each stage are designed in a sequence. This is done so that the minimum length of the shaft can be found out from gear face widths. This is the maximum length of all the shafts. The bending moment distribution on the shaft can now be found out form which the maximum equivalent moment is evaluated. Minimum diameter necessary for the shaft, to keep stresses within safe limit, is determined. Bearings are then selected such that all the requirements are fulfilled. Obtained shaft diameter is then rounded off to the inner diameter of selected bearing. Keys are

every shaft. A check for the possible interference of gears on one shaft with the other shaft is made. If gears are merging, the design is rejected and started again right from design of gears. The layout of gears, key ways on the shaft and gear hub and position of bearings is found out from their dimensions. The data related to dimensions of various elements is stored in a file for further reference.

Lubricating oils recommended [2] for gear speed reducers are as follows

1. SAE # 80
2. SAE # 90
3. SAE # 140
4. SAE # 250

These lubricants are commonly used lubricants for speed reducers with closed casing [2].

2.8 GEAR BLANK DIMENSIONS [2]

The Fig. 2.7 shows a schematic drawing of gear in which the value of different diameters, in terms of diameter of shaft are given.

$$d \approx 0.3\ a \qquad \ldots (80)$$

$$d_1 \approx 1.6\ d \qquad \ldots (81)$$

$$d_2 \approx (D_o - d_1) / 5 \qquad \ldots (82)$$

$$D_1 = (D_o + d_1) / 2 \qquad \ldots (83)$$

$$D_o \approx d_a - 10\ m \qquad \ldots (84)$$

$$\text{rib thickness} \approx 0.3\ b \qquad \ldots (85)$$

where d diameter of shaft cm

Clearences used in the speed reducer schemes

Nomenclature	values
Distance of ratating parts from inner wall of hous.	10 - 15 mm
Distance between adjacent rotating parts	10 - 15 mm
Minimum clearence between gears and inner wall t casing thickness	> 1.2 t
Distance between bearings of over hanging shaft d diameter of shaft	2.5d - 3d
Distance of bearing from wall	5 - 10 mm
Distance between gear and shaft	≥ 20 mm
Width of boss of rotating parts	1.2d-1.5d

table 2.10

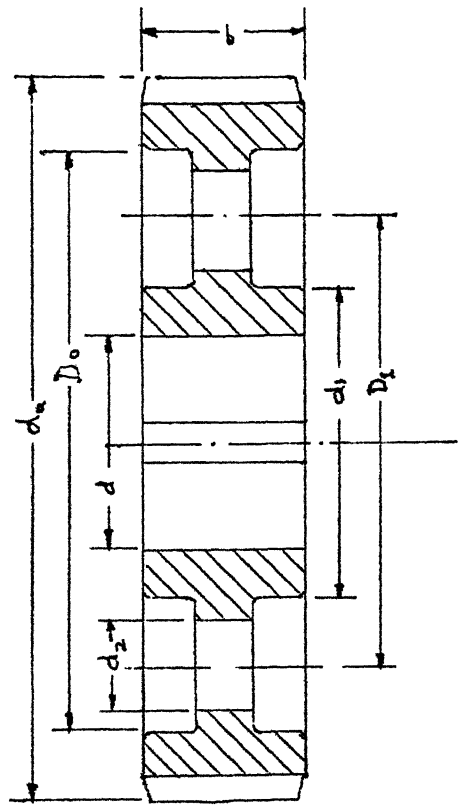


Fig 2.7 Cylinderical gear blank dimensions

a center distance cm

d_1 diameter of hub cm

d_a tip diameter cm

b face width cm

m module cm

Dimensions and clearences required in the speed reducer layout are shown in table 2.10

CHAPTER 3

SYSTEM IMPLEMENTATION

3.1 INTRODUCTION

Implementation of present work has been done in the higher level language 'C' in UNIX environment on APOLLO computer (NEXUS 3000) with one bit plane frame buffer and monochrome monitor. The 'C' language was chosen mainly because of its association with UNIX and provision of a rich set of data structures. Besides these, more efficiency during execution as compared to other higher level languages is achieved due to compact code generated by C compiler. The system is built with the basic idea of designing a mechanical system in front of a terminal, with sufficient flexibility given to the user to change various parameters and observe the results. The arrangement of gears, keys, bearings etc. on the shaft, shape of the shaft with location of key-way and exact contour of gears can be seen in graphic output. The output, related to dimensions and specifications of various components described in the previous chapter, can be stored in a file as well. This chapter describes the salient features of implementation of the system and difficulties faced during the development phase.

3.2 BASIC INTERACTIVE MODEL

Interactive menu based systems are input driven. The

action to be taken is decided by the input. Such types of systems consists of a main infinite loop which controls the working of different modules. In such systems one menu leads to submenus.

The basic Infinite loop consists of three steps

step 1. Wait for an input event to occur

step 2. Change structure or parameters accordingly

step 3. Take the required action

3.2.1 User Interaction

In input driven systems, user is involved at every stage of decision making process. Basic devices used for taking input from the user are key-board and mouse. There are several ways by which input can be taken from the user, from the key-board of the terminal. These can be divided broadly in to two categories. One consists of moving the cursor on the screen to the desired menu and selecting it by pressing a return key. In the other method, strings describing the event are prompted to the user and selection is made by typing in the identifier of the event. This identifier can be a number or a character. A suitable type of error handling is usually incorporated in interactive systems to take care of wrong entry. In this system, the latter strategy of accepting input from the user has been included. This is more simpler to design and implement.

In such systems a word to be input from a prompt string, can be identified by its only-upper-case character. This can be entered either in lower or upper-case. Default

values stored in the system are displayed by enclosing them in < >. Whenever such a construct appears in the prompt string, a different value for the variable can be entered or the same is accepted by merely pressing a return.

3.2.2 Error Handling

In input driven systems, likelihood of wrong input cannot be ignored. A wrong entry, if taken in by the system, can lead to a crash. If the system is small, it can be restarted but if it is not, some sort of error handling is indispensable.

In the present system, input of right type of data item only is allowed. A check for validity of input is incorporated. Infinite loops are included which get exited only if valid data is put in. In this way a valid input is assured in the system.

The basic error handler consists of the following steps

step 1. take input

step 2. check validity of data; if wrong go to step 1.

step 3. check validity of selection; if wrong go to step 1.

step 4. exit the loop.

3.3 STRATEGY OF SYSTEM DESIGN

The system is based on the strategy of modular design. At the highest level, system is divided into two main parts.

i. Mathematical processing part

ii. Graphics part

The graphics portion is kept completely isolated from the

mathematical part. The former is concerned with the calculation of various dimensions required for design of different mechanical components. The graphics is oriented towards the output of results.

The whole program is divided in to several files to facilitate easy handling and maintenance. C functions defined in these files follow the idea of top down structure in design. All the functions are, to a certain extent, written in 20 to 80 lines to minimize the time required for debugging and facilitate easy handling. Separate modules have been designed for various mechanical components.

3.3.1 Selection of data structures

Efficient design of data structures is necessary for programs to have clean structure and efficient working. Before trying to design a data structure for a particular problem, its requirements have to be considered. The possibility of existence of other different structures is then found out and the best suitable for the problem is selected. Sometimes the nature of problem immediately suggests the data structure for it. For example: a space vector can be represented by its three mutually perpendicular components. These components can be stored in a structure with three fields for every component. The vector can be referenced by this structure.

If there are many variables involved in the design of a particular element, the parameters required for the design can be

stored in classified structures i.e., similar type of variables are stored collectively in a structure. These structures are then nested such that every information about that part is available. This becomes more convenient to handle.

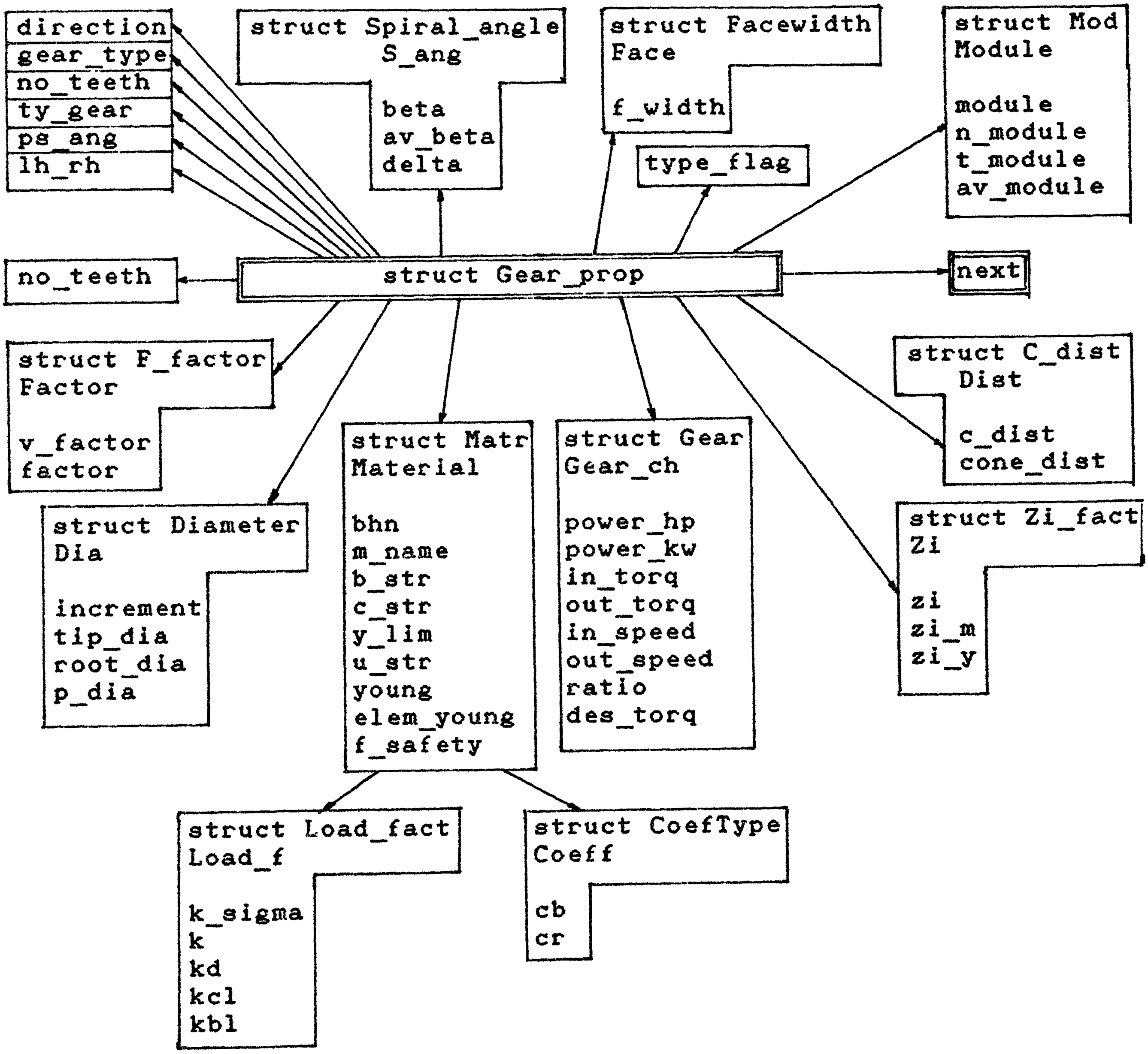
For making a group of similar structures or data items, arrays or linked lists can be used. If these are compared, the lists are more memory efficient than array structure. On the other hand, data in array can be more efficiently processed than lists, if access time is compared. The basic draw back with array structure is its size which is required to be declared before the use. If arrays are used where the size is unpredictable, memory is wasted because of unused cells. In such cases, linked list seems to be a good solution. In this system, linked lists are liberally used wherever the number of entries are not known.

3.4 MATHEMATICAL PROCESSING

The theoretical base required for the design of various mechanical components is explained in detail in the previous chapter. The modules which design the components, with their data structures, are explained in the following subsections.

3.4.1 Design of gears

Three types of gears, explained previously, are designed by the modules spur_design, helical_design and straight_bevel_design for the spur, helical (herringbone) and straight bevel gears respectively. The input to these modules is a pointer to the structure Gear_prop which contains other



Data structure of Gear_prop

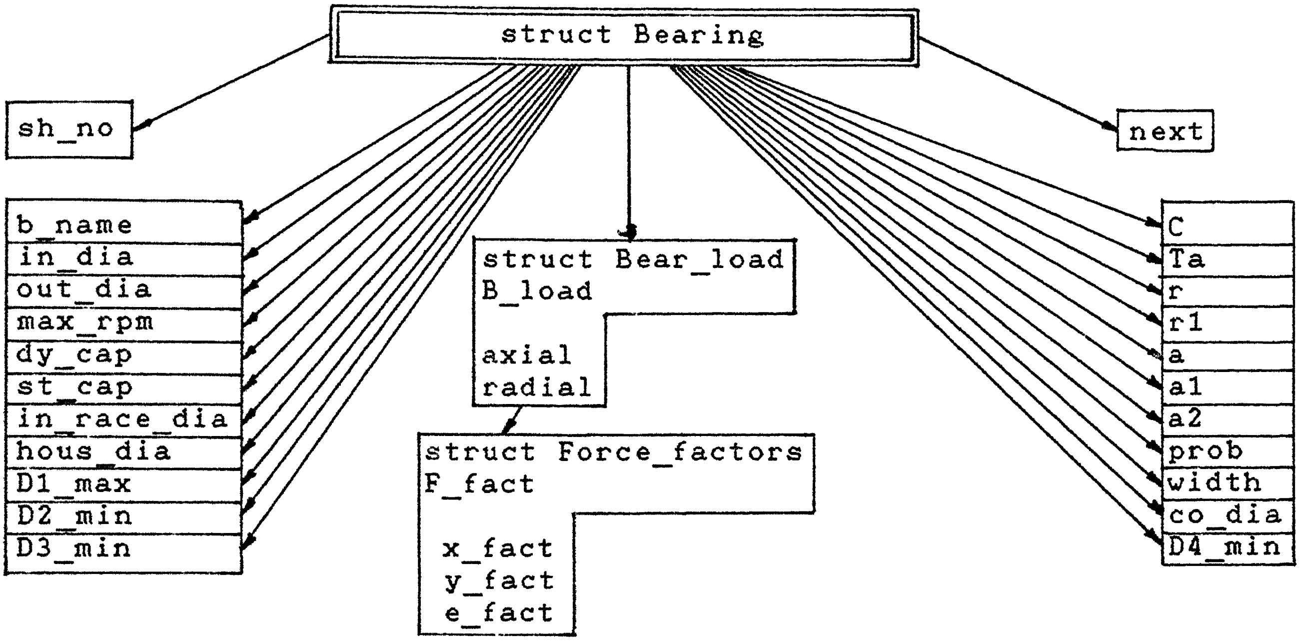
Fig. 3.1

different structures related to various sets of variables. A block diagram showing this is given in Fig. 3.1.

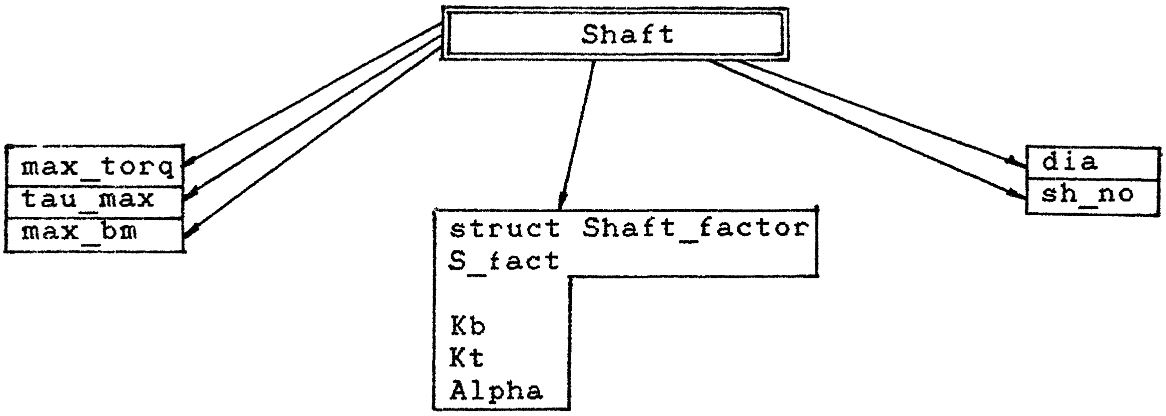
Before entering the routine which designs it, the fields Material and Gear_ch of Gear_prop are filled with the necessary information about material properties and data related to power, speed, etc. The properties of each element, gear or pinion, is stored in different structures and linked to form a link list of two nodes. After the design is over the structure is copied to another structure of same type categorizing shaft-wize i.e. each shaft would contain a gear or pinion or both. Function make_gear_list does this task. The original version is used for making dynamic data base of materials (to be explained later).

3.4.2 Design of shafts

The information necessary for the design of shaft is its material properties and loading condition. Because of gears, a distribution of forces, acting at different locations, exists. q_reaction finds the force exerted by gear tooth because of power transmission. All the force occuring on a shaft because of gears are stored in a list (make_list_force). The distances at which these forces are acting are also stored in a list. The function make_dist_list makes this list. find_reaction computes the reactions at the support points and inserts those distances, at which the reaction is occuring, and reaction forces in the list, whose headers are contained in structure Stage with identifier l_hdr and f_hdr. The inputed material properties are stored in the structure Design_stress_info. app_dia



Data structure of Bearing
Fig. 3.3

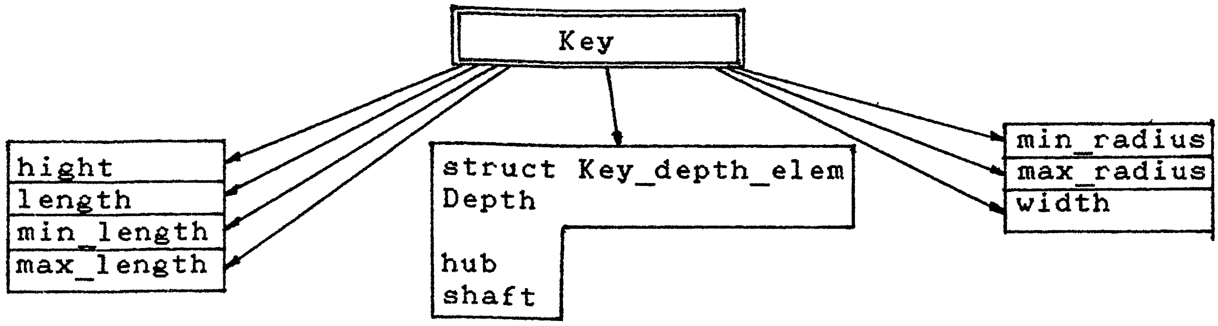


Data structure of Shaft
Fig. 3.2

finds out the minimum diameter necessary for the given working conditions.Structure Shaft contains the information about maximum bending moment (max_bm), maximum torque (max_torq), minimum diameter of shaft necessary(dia) and a structure which contains different factors related to working condition. Pointer to this Shaft is then stored in sh_ptr of Stage. The pointers to Design_stress_info are stored in a list whose header is contained by Prev_mat_info's s_ptr. This is done to update the shaft material's dynamic data base. The data structure of Shaft has been shown in Fig 3.2

Bending moment distribution

The forces can be represented by vectors with three components. The structure Vector contains three components of a spatial vector and a pointer to the next vector. The structure Reaction contains the information about axial, radial and tangential components of tooth force of different type of gears, the planar angle made by the local axis with global axis and a pointer to the next. Function make_long_b_m_list makes a list of moments acting at a regular distance from one end, given the list of forces and distances. Bending moment distribution in two perpendicular planes can thus be found out. The function find_max_mag finds the maximum magnitude of a list of forces or moment vectors, which is required to find critical diameter of shaft.



Data structure of Key
Fig. 3.4

3.4.3 Selection of bearings

The status of load on the bearing is found out by the function find_reaction. The type of bearing, to be employed at the specified distance, is inputed by user and is coded in program by function find_bear_type. sel_bear_x_y_fact finds out the X and Y factors, explained earlier, for the bearing selected by the user. The function bearing_selection finds out the bearing, with type specified by user, of inner diameter just greater than or equal the diameter of shaft. make_bear_list makes a list of Bearing finally selected by the user for both ends of the shaft. The header of this list is stored in the bear_ptr of structure Stage. To search the data base of bearing data, the record number in the form of line number is found out with the help of get_add. This utilizes a key table for searching with respect to the diameter or dynamic capacity, representing those records. The actual data base is read by get_specifications and stored in the structure Bearing. Fig 3.3 shows the data structure of Bearing.

3.4.4 Design of keys

Keys can be designed or selected. The function design_key either designs a key or finds a suitable key, with reference to the diameter of shaft, depending upon the select_flag. This select_flag value is selected by the user, while designing the same. The information related to the material is contained by Design_stress_info. The final dimensions, after the design is over, are stored in the structure Key. A list of Key, for every

shaft corresponding to each gear element is made, and its header is stored in the key_ptr of Stage. The pointer to the Design_stress_info is stored in the list whose header is contained by Prev_mat_info's k_ptr. The dynamic data base of key material is updated this way. Fig. 3.5 gives an idea about the structure in which the list header is stored. Fig. 3.4 shows the data structure of Key.

3.4.5 Design of sleeves

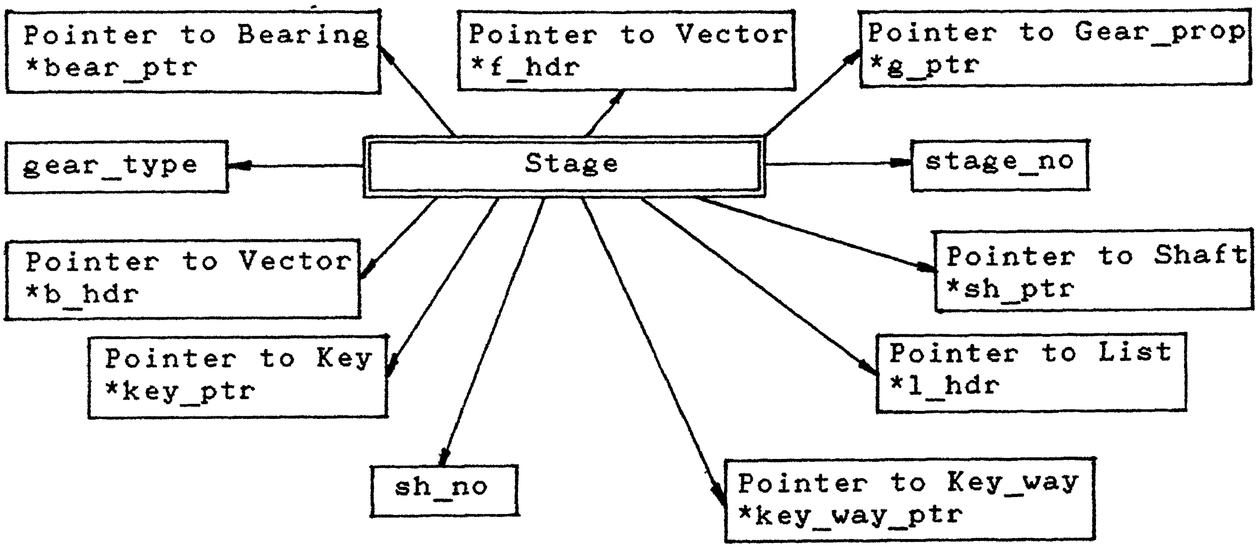
The only variable related with design of sleeve is its thickness. The inner diameter is equal to the shaft diameter. The allowable compressive stress is inputed by the user. The sleeve thickness is limited by the bearing inner race diameter. Hence one strategy can be used which checks the induced compressive stress in the sleeve material, with assumed thickness of the difference of inner race radius and bearing inner radius. The length of sleeve is decided by the distance between gear element and bearing face.

3.4.6 Critical casing thickness

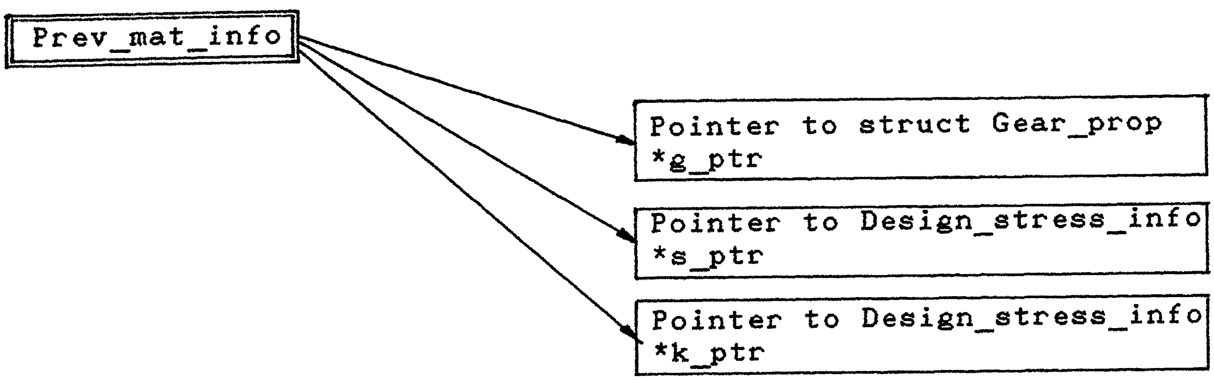
The critical tensile load acting on the casing rib is the maximum of all loads between two adjacent bearings. The function find_critical_casing_th calculates the critical load and finds out the minimum thickness necessary.

Need of Data Base in the system

The data base is needed in this system for the material of gears, shafts and keys. Two types of data bases have been



Data structure of Stage
Fig. 3.6



Data struture of Prev_mat_info for dynamic data base
Fig. 3.5

incorporated in the system. One is dynamic in nature and the other static in nature. The static data base is stored in the system and can be referred at will. Dynamic data base is maintained to store material properties, which the user has inputed earlier in design process of gears, shafts and keys. As shown in the Fig. 3.5, the structure Prev_mat_info contains the headers of the lists of structures Gear_prop and Design_stress_info. Pointers s_ptr, k_ptr and g_ptr are the headers of the list of the material of shaft, key and gear respectively.

3.5 IMPLEMENTATION OF GRAPHICS

APOLLO system provides two graphics packages which are essentially collections of graphic routines. These are as follows

1. GMR - Graphics Metafile Resources
2. GPR - Graphics Primitive Resources

GMR, as the name suggests, provides the metafile facility, by which the graphic objects can be stored internally and retrieved back later. The normalization transformation used for display is done automatically. GPR, on the other hand, is the fundamental one and hence stands one level lower than GMR. So it is more efficient than GMR from the execution speed point of view. Routines are easy to work with. The normalization transformation required for display is provided by user. In the present work, GPR is used mainly because of its simplicity and efficiency.

3.5.1 Salient features of GPR

GPR provides the ability to create, display and store the whole or a part of display buffer in a file. The basic operation exists in two modes; namely the direct mode and borrow mode. In borrow mode, full screen is borrowed, irrespective of working window size, from the display manager of the computer. The direct mode utilizes the default window size for display. Borrow mode is felt more convenient than the other. The other basic graphics features are almost similar to GKS (Graphics Kernal System), which is one of the standards available in graphic routine design. The philosophy of working of these packages is as follows. Choice of background color is optional i.e., the background can be kept white of black, but in borrow mode background is made black by the computer. When the graphics is initialized, key board keys and buttons in mouse are disabled by the computer. They can be enabled by giving explicit instructions in the program.

3.5.2 Requirement of graphics

After the design phase is over, the component designed has to be manufactured. To have the exact dimensions of the component, final drawings are necessary. The data related to component can either be given in the form of a table or scaled drawings. Graphics plays an important role here. Components can be drawn with suitable scale with exact curvature requirements so that the post-processing becomes easy. A visual feedback, immediately after the design is over, can be in a way,

advantageous to the designer. Even the hard copy output can be sent directly to the manufacturing section for further processing. This process eliminates one stage of conventional design of mechanical components. Graphics output is required only for those parts which are required to go in the machine shop. Standard components can be referenced through the label, may be in the form of a code number signifying it and hence every information related to dimensions, material, etc., are known at priori.

3.5.3 Scaled drawing of gears

Gears are drawn to scale to the exact requirements. As pointed out earlier, the tooth profile is considered to be of involute type. Points lying on the fillet portion of the gear tooth are found out as given in the last chapter. In order to draw the whole gear as a single element following is done. The tooth is broken into three parts. One consists of involute curve, second the fillet portion (trochoid) and the third consisting of a circular arc. Three lists representing the curve, are made and appended to form a list for half tooth. A mirror image (x coordinate negated) is then taken to get the other half. These two lists are then appended to form a single list representing a complete tooth. Points lying on the tooth which, are in the form of a list, are drawn. A rotation transformation on the list is then performed to get next tooth. This process, when repeated for the total number of tooth, gives a complete gear.

CHAPTER 4

PERFORMANCE AND RESULTS

4.1 PERFORMANCE OF THE SYSTEM

The system is developed in such a way that the user can take decisions at critical stages of design. Values of the parameters such as Ψ, Ψ_M, Z1, etc., can be changed and their effect seen immediately on the design.

While taking the input of material data for gears, shafts, keys and sleeves in the system, a dynamic data base is maintained. This strategy helps in selecting those materials which have been entered previously during design.

Final drawing of the gears of each stage and the location of keyway in the shaft can be displayed. Dimensional data of these components are stored in a file so that these can seen for further details.

4.2 TRIAL RUN

The following problem has been considered as an example.

To design gears, shafts, keys, bearings, sleeves for an application where mild shocks are present in the load. Spur gears are to be used for power transmission. The first and second stage reduction is 2 and 2.2 respectively. Total power transmitting

through the system is 3 hp and the output speed required for the application is 140. Following is the information about various other data.

Expected life	5 years
use in hrs/day	8 hrs
Service factor	1.2 (mild shock)
probability of survival of each bearing 95 %	
Gear Material	CIgrade25
Pinion Material	C10
Shaft Material	40Cr1Mo28
Key Material	C40
Sleeve Material	C40

4.3 RESULTS OF TRIAL RUN

Results obtained in trial run are given below.

FIRST STAGE

Gear and Pinion material properties kg/sq.cm

Element	Material	BHN	σ_u	σ_y	E	σ_{-1}	$[\sigma_b]$
Pinion	C10	127	3200.0	2100.0	$2.1\ 10^7$	1825.0	1055.0
Gear	CIgrade25	197	2500.0	1850.0	$2.1\ 10^6$	1125.0	650.8

Gear and Pinion dimensions

Element	Z	b	m	y	d_p	d_r	d_t
Pinion	18	1.62	0.3	0.38	5.4	4.65	6.00
Gear	36	1.62	0.3	0.45	10.80	10.05	11.40

First Stage Properties

In_speed	Out_speed	Ratio	Power	In_torq	Out_torq	Center_dist
616.0	308.0	2.0	3.0	348.8	697.6	8.10

Shaft and bearing properties

Shaft Material 40Cr1Mo28

Diameter of first shaft 2.0 cm

Bearing Left End SKF 6204

Bearing Right End SKF 6004

Shaft material 40Cr1Mo28

Diameter of Second shaft 3.0 cm

Bearing Left End SKF 6006

Bearing Right End SKF 6006

Key dimensions

First shaft (diameter 2.0 cm)

#	width	length	hight	max_r	min_r	depth_s	depth_h
1.	0.6	2.12	0.6	0.04	0.03	0.35	0.28

Second shaft (diameter 3.0 cm)

#	width	length	hight	max_r	min_r	depth_s	depth_h
1.	0.8	2.12	0.7	0.04	0.03	0.40	0.33
2.	0.8	2.78	0.7	0.04	0.03	0.40	0.33

SECOND STAGE

Gear and Pinion material properties kg/sq.cm

Element	Material	BHN	σ_u	σ_y	E	σ_{-1}	$[\sigma_b]$
Pinion	C10	127	3200.0	2100.0	$2.1\ 10^7$	1825.0	1055.0
Gear	CIgrade25	197	2500.0	1850.0	$2.1\ 10^6$	1125.0	650.8

Gear and Pinion dimensions cm

Element	Z	b	m	y	d_p	d_r	d_t
Pinion	18	2.28	0.4	0.38	7.2	6.25	8.00
Gear	39	2.28	0.4	0.46	15.60	14.60	16.40

Second Stage Properties

In_speed	Out_speed	Ratio	Power	In_torq	Out_torq	Center_dist
308.0	154.0	2.2	3.0	697.6	1395.2	11.4

Shaft and bearing properties

Shaft Material 40Cr1Mo28

Diameter of first shaft 3.0 cm

Bearing Left End SKF 6006

Bearing Right End SKF 6006

Shaft material 40Cr1Mo28

Diameter of Second shaft 6.0 cm

Bearing Left End SKF 6012

Bearing Right End SKF 6012

Key dimensions cm

First shaft (diameter 3.0 cm)

#	width	length	hight	max_r	min_r	depth_s	depth_h
1.	0.8	2.12	0.7	0.04	0.03	0.40	0.33
2.	0.8	2.78	0.7	0.04	0.03	0.40	0.33

Second shaft (diameter 6.0 cm)

#	width	length	hight	max_r	min_r	depth_s	depth_h
1.	1.8	2.78	1.10	0.05	0.04	0.70	0.44

Critical length of the shaft = length of second shaft

Length of gear hub = 2.12 cm

Length of pinion hub = 2.78 cm

Wall clearence = (1.0 + 1.0) cm

Bearing clearence from wall = (0.5 + 0.5) cm

Assumed width of bearing = 3.0 cm

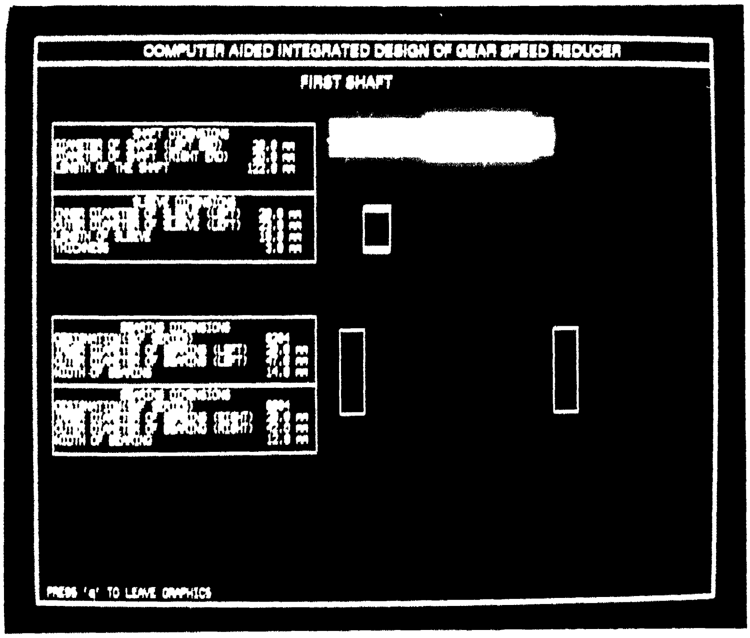
Clearence between rotating gears = 1.0 cm

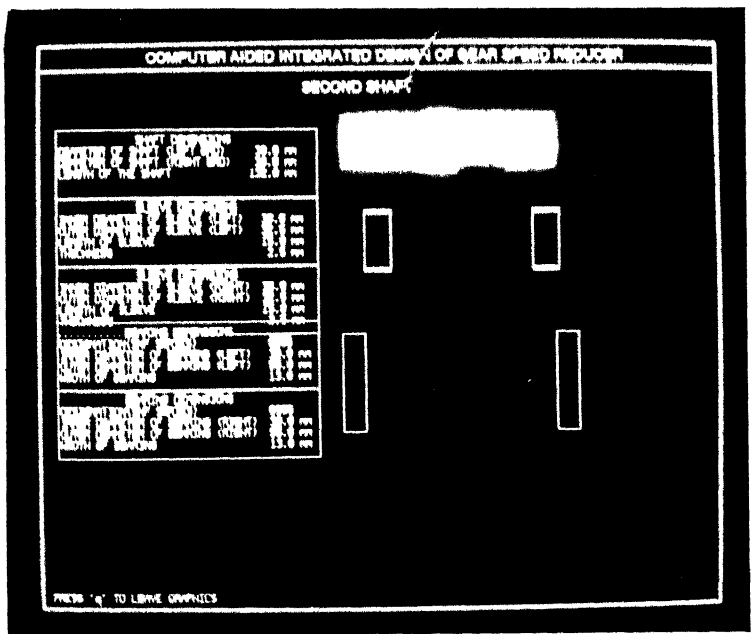
Total length = 11.9 cm

shaft #	sleeve_id	Length_of_sleeve	inner_diameter	outer_diameter
1	left	1.8	2.0	2.6
2	left	1.85	3.0	3.5
	right	1.85	3.0	3.5
3	right	1.6	6.0	6.6

Speed ratio after design 4.33

percent deviation -1.52 %


COMPUTER AIDED INTEGRATED DESIGN OF GEAR SPEED REDUCER
FIRST SHAFT
PRESS 'q' TO LEAVE GRAPHICS


COMPUTER AIDED INTEGRATED DESIGN OF GEAR SPEED REDUCER
SECOND SHAFT
PRESS 'q' TO LEAVE GRAPHICS

CHAPTER 5

CONCLUSIONS AND DIRECTIONS FOR FURTHER WORK

5.1 CONCLUSIONS

In the present work, design of gears, shafts, keys, sleeves and selection of antifriction bearings have been covered to form a basic gear speed reducer.

Design of lubrication system and casing have not been considered in detail. However, complete design of these is beyond the scope of this work.

The computer system (APOLLO, NEXUS 3000) on which the implementation has been done, does not provide necessary support, for taking hard copy output of graphic objects, at present. However, the same can be obtained if suitable device drivers for plotter and printer are written.

5.2 DIRECTIONS FOR FURTHER WORK

Three types of gears have been included as the basic torque transmitting members in this system. The other types which can be considered are worm, spiral bevel, etc. Thrust bearings are necessary if worm gears are included in the system. The relevent data can be taken from [2] and stored in the system.

Only two gears per shaft have been considered in the

system. A complex gear speed reducer can have more than two gears per shaft. If combinations of various gears are considered, design of a rule based system for selecting right type of combinations of above gear types may be required.

Stress analysis of gear tooth can be done from which optimal tooth size can be found out. FEM (Finite Elements Method) can be used in the design process itself to find out optimal dimensions of gear for the safe stress level. FEM module for this, would require a large memory. It will be a major factor while designing.

As pointed out earlier, other parts which remain to be designed are casing and lubrication system. The casing design would involve interactive ways of constructing surfaces according to the requirements.

Lubrication system in the speed reducer would consist of options for method of lubrication, viz. force feed, splash, etc. The layout of the lubricant carrier lines in the box, the design of pump for optimum power consumption, lubricant selection for the given working environment, etc., would be the major factors for design.

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APPENDIX 1

USER MANUAL

This manual describes the sequence of events and usage of prompts appearing on the screen. Prompts are self explainatory; however, a brief description of the data has been given wherever necessary.

A brief description about files necessary for this system is as follows

1.	g_struct.h	header file; contains the data structure.
2.	macro.h	header file; contains the macro definitions.
3.	*.c	C Source code files.
3.	*.dec	Function declairation files.
4.	*.dat	Data base files.
5.	*.doc	Documentation on the system.
6.	makefile	UNIX project make; makefile facility.
7.	reducer	The speed reducer

When the system is run by executing the file reducer a brief introduction about gear speed reducer appears on the screen. Data related to global inputs are then asked. These are as follows

(1)

Is the machine Unidirectional? y/n :

The machine (speed reducer) can run in only one direction or both the directions, depending upon the application. This information is asked to determine design bending strength of the gear and pinion material.

(2)

Enter file name in which out put is to be stored <stdout>
:

The output of the program can be stored in a file. Type in the name of file in which the output is to be stored. Terminal is kept as the default.

(3)

Enter the expected life of the system in YEARS <5.00> :

The life of the system in years, is to be entered.

(4)

Use in number of Hours/Day <8.00> :

The machine usage in hours per day is required. The information about total number of hours of operation is required to find the life in million revolutions and dynamic capacity needed for the bearing.

(5)

Enter service factor (1 < x < 1.5) < 1.20> :

The service factor in the range 1 to 1.5 is required. It depends on the the severeness of application. If the application involves dynamic loads frequently, a higher factor should be chosen. For the service where operation is relatively smooth, default value can be selected.

(6)

Probability of survival of each Bearing < 0.90> :

Probability of survival of every bearings in the system is required. Bearings can have probabilities, theoretically, between 0 and 1. However, the values in the real situation vary between 0.5 and 0.99 depending on the application.

(7)

Enter the choice for gear combination to be used

1. SPUR gears only
2. HELICAL(HERRINGBONE) gears only
3. BEVEL gears only
4. BEVEL and SPUR gears
5. BEVEL and HELICAL gears

enter choice <1> :

The combination of gears to be used in the system can be

selected by entering the number designating it.
(8)

Input no. of Stages <2> :

A number, signifying total number of stages to be incorporated in the system, is required.
(9)

Individual Stages Ratio? y/n :

The question is whether you want to enter the stage ratios individually for each stage? If yes, stage ratios are asked starting from the first stage or if no, the overall speed ratio will be divided equally amongst all stages.
(10)

Power/Input_torque/Output_torque <Power> :

If power transmitted through the system is known, just <return> or enter the character in upper case or lower case for any other entry.

following is the list of sub-menus for the entered character.

<return> or 'p' : Horsepower/Kilowatt/Watt <Horsepower>:
- <return> or 'h' : Horsepower :
- 'k' : Power in KW :
- 'w' : Power in watt :

'i' :Input Torque in N-m/Kg-m/kg-Cm <Kg-m> :
- <return> or 'k' : Torque (Kg-m):
- 'n' : Torque (N-m):
- 'c' : Torque (N-m):

'o' : Output Torque in N-m/Kg-m/kg-Cm <Kg-m> :
- <return> or 'k' : Torque (Kg-m):
- 'n' : Torque (N-m):
- 'c' : Torque (N-m):

(11)

if (9) is 'n'.

Speed_ratio/speeD :
- 's' : Speed ratio :
- 'd' : Input_speed/Output_speed :
  - 'i' : Input speed in rpm :
  - 'o' : Output speed in rpm :

if (9) is 'y'

Input_speed/Output_speed :

'i' : Input speed in rpm :

'o' : Output speed in rpm :

(12)

First shaft rotating anticlockwise? y/n :

To find the direction of rotations of all gears, the direction of first gear is asked. The design of gears follows this command.

(13)

Enter the material of the PINION

RETURN for the list of Materials already selected:

If a material is to be enterd with its properties, just enter the name signifying the material. If the static or dynamic data base is desired just <return>

Similar type of prompt will appear for GEAR material selection.

(14)

Enter your choice or RETURN for data-base:

If any material has been selected previously, the properties are displayed. Any of it can be selected by its number. If system data base is desired, hit <return>.

(15)

Enter the type of the material

1. Forged steel
2. Cast steel
3. Alloy steel
4. Cast Iron

Enter Your choice < 3 >:

Enter the desired choice.

(16)

1> Enter the factor of safety in bending (2.0 - 2.5) :

The factor of safety for finding the design bending strength is required. The factor of safety for heat treated and hardened surface of gear material is near to 2.5. It depends on the heat treatment of material. The factor of safety can be selected from the table comming on the screen just before this

command.

(17)

Is it O.K. y/n:

Enter the response y or n. If n is entered, the previous value inputed is shown in < >. A new value can be entered at this stage.

(18)

1> Enter the stress concentration factor (1.0 - 1.2):

Stress concentration factor for fillet is required and can be entered in the range.

(19)

Enter the load concentration factor k (1.0 - 1.2) < 1.10> :

The load concentration factor depends on the following

1. Location of gear and pinion relative to bearing
2. Rigidity of shaft
3. Rigidity of teeth of gear element
4. The ratio of face width to center distance.

(20)

Enter the dynamic load factor kd (1.0 - 1.2) < 1.10> :

1. Surface hardness of tooth
2. Degree of accuracy in machining
3. Pitch line velocity
4. Stiffness of the teeth.

(21)

Want to change material properties y/n :

If the properties of material is to be changed, after inputting the same, one more chance is given here.

(22)

number of teeth in pinion <18> :

Number of teeth in pinion is required. Default is kept at 18 which is the minimum number of teeth to avoid undercut.

(23)

select module < 0.30> :

Module should be selected from the list printed above this command. Default shows the minimum module calculated in the

routine. The list printed gives the modules in order of preference.

Bending and surface stress levels are printed along with the dimensions of gear and pinion.

(24)

Whether the design is acceptable or not y/n :

If the stress levels are not exceeding the designed strength(displayed) 'y' should be entered. If any modification in any value or material properties are required, enter 'n'.

(24)

want to change material properties y/n :

if 'y' :

prompts related to material properties appear on the screen

if 'n' :want to change # of teeth of pinion y/n :

'y' : Enter pinion # of teeth <18> :

if gear type is spur or helical or herringbone

'n' : Enter new value of (b/a) < 0.20> :

(25)

Want to edit distances already inputed y/n :

The distances of gears from left end of shaft are shown. If their position is to be changed, enter 'y'.

(26)

Enter the material of SHAFT

RETURN for the list of Materials already selected:

This event is similar to event (13)

<return> : event (14)

string :1> Enter the allowable SHEAR STRESS of the material

event (17)

1> Enter the allowable COMPRESSIVE STRESS of the material

event (17)

(27)

Name of Material :

Input the name of material. Event (17) follows it.

(28)

1. DEEP GROOVE BALL BEARING
2. ANGULAR CONTACT BALL BEARING
3. ROLLER BEARING
4. TAPER ROLLER BEARING
5. SPHERICAL ROLLER BEARING

enter choice <1> :

Any of the choices can be selected for the application. The axial and radial force are shown in a tabular form.

(29)

if '1' :

1. 60 SERIES
2. 62 SERIES
3. 63 SERIES
4. 64 SERIES

Enter choice <1> :

This is the series available for Deep groove Ball bearing.

if '2' :

1. 72 SERIES
2. 73 SERIES
3. 33 SERIES

Enter choice <1> :

This is the series available for Angular contact Ball bearing.

if '3' :

1. 22 SERIES
2. 23 SERIES

Enter choice <1> :

This is the series available for Roller bearing.

if '4' :

1. 322 SERIES
2. 323 SERIES

Enter choice <1> :

This is the series available for Taper Roller bearing.

if '5' :

spherical roller bearing is selected.

(30)

Want to reselect choice y/n :

if 'y', the event (28) is executed.

(31)

Want to have both the bearings of same type and capacity? y/n :

If both the bearing are to be same enter 'y'. The bearing with larger dynamic capacity will be selected for both the ends.

(32)

Enter length of key in cm <   3.20> :

The length of key is calculated with respect to the face width of gear. However it can be replaced by another value.

(33)

Want to Select key y/n :

Keys can be selected or designed. if entered 'n', the key will be designed and the dimension otained will be compared with the standard dimensions available. Another question about selecting the standard dimensions will be asked if the obtained dimensions are lesser than standard.

(34)

Enter the material of KEY

RETURN for the list of Materials already selected:

The event is similar to (26)

(35)

Want to see gear profile y/n :

(36)

if event (35) is 'y' : enter the stage # <1> :

The exact gear pair contour is displayed on the screen. Any stage can be observed this way.

(36)

want to see other gears in diff. stages y/n :

if 'y' : event (36)

(37)

Enter width of support web <   3.00> :

Thickness of casing's support web, which usually takes the load is calculated. The critical load acting on this web is printed before this prompt.

(38)

Enter tensile strength of casing material < 3000.00> :

In material properties only tensile strength is required.

(39)

LUBRICANTS RECOMMENDED FOR THIS APPLICATION ARE AS FOLLOWS

1. SAE # 80
2. SAE # 90
3. SAE # 140
4. SAE # 250

choose one from the above <1> :

These are the four suitable lubricants for this application.

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